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グ冷凍エンジンの放熱効率を高め、スターリング冷凍エンジンの冷凍能力を十分に発揮させられるようにすることにある。また、スターリング冷凍エンジンの高温部の発する熱を冷却庫の機能向上に役立て、同時に電力消費量を低減できるようにすることにある。

上記目的を達成するため、本発明では冷却庫を次のように構成する。すなわちスターリング冷凍エンジンにより庫内冷却を行う冷却庫において、前記スターリング冷凍エンジンの高温部に設けた高温側熱交換器と、庫外環境に放熱を行うための放熱用熱交換器とを備え、ループ状サーモサイフォンを構成してなる第１高温側冷媒循環回路と、前記高温側熱交換器と、ドレンの蒸発促進と冷却庫壁の結露防止の少なくとも一方に利用する熱交換器と、循環ポンプとを備え、前記高温側熱交換器の下部から該高温側熱交換器内の冷媒を前記熱交換器に送り出すように構成してなる第２高温側冷媒循環回路と、を備えるものとする。

この構成によれば、スターリング冷凍エンジンの高温部の熱を庫外に放熱する第１高温側冷媒循環回路を設けることにより、高温部の熱を安定して放熱できる。第１高温側冷媒循環回路は、高温側熱交換器と放熱用熱交換器との間にループ状サーモサイフォンとして構成されているから、高温側熱交換器より、人工的なエネルギーを使用することなく熱をくみ出すことができる。

そして第２高温側冷媒循環回路では、高温側熱交換器の下部から該高温側熱交換器内の冷媒を循環ポンプで熱交換器に送るから、高温部の熱をドレンの蒸発促進と冷却庫壁の結露防止の少なくとも一方に確実に利用することができる。これによりドレンのメンテナンスフリー化を図ることができる。また電熱ヒーターを用いずに冷却庫壁の結露を防止することができ、冷却庫の機能あるいは使い勝手が向上するとともに、加熱を電熱ヒーターにより行う場合に比べ、消費電力を抑えることができる。

またドレン水や結露懸念部から周囲環境より温度の低い冷熱を回収してスターリング冷凍エンジンの高温部を冷却するので、放熱システム全体の放熱効率が向上する。スターリング冷凍エンジンのＣＯＰも向上し、冷却庫の電力消費量を低減できる。

そして第2高温側冷媒循環回路内の循環ポンプは、常時運転の必要はなく、ドレンの蒸発促進や扉周辺の結露防止が必要となったときのみ運転すればよいから、循環ポンプの電力消費を節約し、循環ポンプの稼働寿命を延ばすことができる。また扉周辺を必要以上に長く加熱しないので、冷却庫の熱負荷を低減し、消費電力を抑制することができる。

また本発明では、前述のように構成される冷却庫において、前記高温側熱交換器を2個設けると共に、前記第1高温側冷媒循環回路と前記第2高温側冷媒循環回路を、前記2個の高温側熱交換器のそれぞれに対して互いに並列に接続する。

この構成によれば、高温側熱交換器を2個設けるとともに、第1高温側冷媒循環回路と第2高温側冷媒循環回路を、2個の高温側熱交換器のそれぞれに対して互いに並列に接続するから、どちらの高温側熱交換器を取り上げても高温側冷媒循環回路が複数個確保されることになり、回路閉塞による冷媒循環停止といった事態を回避しやすくなる。

そして、2個の高温側熱交換器の両方より、第1高温側冷媒循環回路と第2高温側冷媒循環回路に冷媒の供給が行われ、2個の高温側熱交換器の両方に対し、第1高温側冷媒循環回路と第2高温側冷媒循環回路から冷媒が還流するものとしたから、2個の高温側熱交換器をいずれも外部への熱供給に関与させることができる。

また本発明では、前述のように構成される冷却庫において、ドレンの蒸発促進のために設けられる熱交換器と、冷却庫壁の結露防止のために設けられる熱交換器とを並列接続し、それぞれの熱交換器に弁を設けて前記第2高温側冷媒循環回路を形成する。

この構成によれば、ドレンの蒸発促進のために設けられる熱交換器と、冷却庫壁の結露防止のために設けられる熱交換器とを並列接続したから、冷媒の流動抵抗を低くできる。冷媒の流動抵抗が低いので、循環ポンプを用いる場合、その消費電力を大幅に削減できる。またそれぞれの熱交換器に弁を設けたので、その時点で冷媒を流す必要のない側の熱交換器は冷媒の流れを止めることができ、循環ポンプの負荷を減らすことにより、その消費電力を削減できる。

また本発明では、前述のように構成される冷却庫において、前記高温側熱交換器内の冷媒は気液二相であるものとする。

この構成によれば、冷媒を気液二相の形で用いるから、冷媒の蒸発・凝縮という、潜熱が熱交換に利用されることになり、熱抵抗を小さく抑えることができ、放熱効率が高まる。これにより熱交換効率が飛躍的に高まり、スターリング冷凍エンジンの効率が向上し、消費電力を低減できる。

図面の簡単な説明

図１は冷却庫の断面図である。

図２は本発明冷却庫の第１実施形態を示す配管構成図である。

図３は本発明冷却庫の第２実施形態を示す配管構成図である。

図８は本発明冷却庫の第３実施形態を示す配管構成図である。

図９は本発明冷却庫の第４実施形態を示す配管構成図である。

図１３は本発明冷却庫の第５実施形態を示す配管構成図である。

図１４は本発明冷却庫の第６実施形態を示す配管構成図である。

また循環ポンプ６４は第２高温側冷媒循環回路６０の最上流部に配置されているので、第２高温側熱交換器６１から循環ポンプ64までの管路抵抗が少なく、冷媒はスムーズに循環ポンプ６４に流れ込む。循環ポンプ６４に冷媒を供給する管路の抵抗が大きいと、循環ポンプ６４の吸込側にキャビテーションが生じて冷媒が不必要に蒸発し、循環効率を損なうことがあるが、このように循環ポンプ６４が第２高温側冷媒循環回路６０の最上流部に配置されていれば、そのような事態を避けることができる。

気液二相に関して言えば、第２高温側冷媒循環回路６０において、熱交換部６２、６３でドレン処理と結露防止を行うあたりでは冷媒が液相のみであっても構わない。その冷媒が第２高温側熱交換器６１に還流した時点では、その還液と冷媒蒸気との潜熱熱交換となるため、ここで高い熱交換効率が得られる。

続いて、第２実施形態以下の実施形態を図３以下の図に基づき説明する。図３～図１４はいずれも配管構成図であり、そこに示された配管が図１の冷却庫１の中で実現されているものとする。第１実施形態と共通する構成要素については第１実施形態の説明で使用した符号をそのまま使用し、説明は省略する。

本発明冷却庫の第２実施形態を図３に示す。ここではドレンの蒸発促進のための熱交換部６２と冷却庫壁の結露防止のための熱交換部63とを並列接続し、この並列接続構造を第２高温側熱交換器６１及び循環ポンプ64に直列接続する。循環ポンプ６４はここでも第２高温側冷媒循環回路６０の最上流部に配置される。そして前記並列接続構造の内部において、熱交換部６２の上流側に弁６５を直列接続し、熱交換部６３の上流側に弁６６を直列接続する。

上記構成によれば、熱交換部６２、６３の箇所における冷媒の流動抵抗が第１実施形態の約半分になり、循環ポンプ６４の消費電力を大幅に削減できる。また熱交換部６２、６３に弁６５、６６を組み合わせたので、ドレンの蒸発促進と冷却庫壁の結露防止のいずれかが必要でなければ、必要でない側の弁を閉じて冷媒の流動を止めることができる。循環ポンプの負荷を減らすことにより、循環ポンプ６４の消費電力をさらに削減できる。

結露防止のため必要なとき以外は弁６６を閉じることとすれば、扉１４、１５、１６の周辺が必要以上に長く加熱されることがなくなる。これにより冷却

室１１、１２、１３の熱負荷を低減し、消費電力を抑制することができる。

熱交換部６２、６３のそれぞれに専用の弁を設けるのでなく、共通の三方弁を設け、この三方弁の切り替え操作により「熱交換部６２、６３の両方に冷媒が通る」「熱交換部６２だけに冷媒が通る」「熱交換部６３だけに冷媒が通る」の３状態を選択するようにすることもできる。また自動制御を容易にするため、弁は電磁弁としておくのがよい。

なお第1高温側冷媒循環回路５０と第２高温側冷媒循環回路６０を流れる冷媒はいずれも気液二相である。

本発明冷却庫の第３実施形態を図８に示す。第３実施形態は第２実施形態に対し高温側熱交換器が単一型になっている点が異なる。すなわち本実施形態では単一型の高温側熱交換器７１がスターリング冷凍エンジン30の高温部に取り付けられている。高温側熱交換器７１の内部には多数のフィンが設けられ、冷媒との間で効率よく熱交換を行えるようになっている。

この高温側熱交換器７１を含む形で、第１高温側冷媒循環回路５０と第２高温側冷媒循環回路６０が構成されている。すなわち高温側熱交換器７１は第１高温側冷媒循環回路５０と第２高温側冷媒循環回路６０の双方に共通の高温側熱交換器であり、この共通の高温側熱交換器７１に第1高温側冷媒循環回路５０と第2高温側冷媒循環回路６０が互いに並列に接続された形になっている。

本発明冷却庫の第４実施形態を図９に示す。湿度の高い環境にあってはドレンの蒸発促進と冷却庫壁の結露防止を休みなく行わねばならないが、第４実施形態の配管構造はこのような場合に適するものである。

第４実施形態は第１実施形態に対し高温側熱交換器が単一型になっている点が異なる。すなわち本実施形態では単一型の高温側熱交換器７１がスターリング冷凍エンジン３０の高温部に取り付けられている。高温側熱交換器71の内部には多数のフィンが設けられ、冷媒との間で効率よく熱交換を行えるようになっている。

この高温側熱交換器７１を含む形で、第１高温側冷媒循環回路５０と第２高温側冷媒循環回路６０が構成されている。すなわち高温側熱交換器７１は第１高温側冷媒循環回路５０と第２高温側冷媒循環回路６０の双方に共通の高温側熱交換器であり、この共通の高温側熱交換器71に第１高温側冷媒循環回路５０と第２高温側冷媒循環回路６０が互いに並列に接続された形になっている。

上記構成によれば、第２高温側冷媒循環回路６０の配管構造が簡単で、組立工数が少なくて済むというメリットがある。

熱交換部６２、６３の位置を逆転し、先に冷却庫壁を加熱し、次いでドレンパン２６を加熱するようにしてもよい。

本発明冷却庫の第５実施形態を図１３に示す。第５実施形態は、第２実施形態の構成を次のように変更したものである。すなわち第２実施形態の場合、第１高温側熱交換器５１は第１高温側冷媒循環回路５０に専属し、第２高温側熱交換器６１は第２高温側冷媒循環回路６０に専属していた。第５実施形態では、第１高温側熱交換器５１と第２高温側熱交換器６１の両方を、第１高温側冷媒循環回路５０と第２高温側冷媒循環回路６０で共通に使用する。

図１３に見られるように、第１高温側冷媒循環回路５０の冷媒配管は第１高温側熱交換器５１と第２高温側熱交換器６１の両方から並列に出、途中で合流して放熱用熱交換器５２に入る。放熱用熱交換器５２を出た冷媒配管は途中で分岐し、並列をなして第１高温側熱交換器５１と第２高温側熱交換器６１に戻る。

第２高温側冷媒循環回路６０の冷媒配管は第１高温側熱交換器５１と第２高温側熱交換器６１の両方から並列に出、途中で合流して循環ポンプ64に入る。ドレンの蒸発促進のための熱交換部62と冷却庫壁の結露防止のための熱交換部６３の並列接続構造を出た冷媒配管は途中で分岐し、並列をなして第１高温側熱交換器５１と第２高温側熱交換器６１に戻る。

別の言い方をすれば、第１高温側冷媒循環回路５０と第２高温側冷媒循環回路６０とは、第１高温側熱交換器５１と第２高温側熱交換器６１のそれぞれに対して、互いに並列に接続されている。

上記構成により、第１高温側熱交換器５１と第２高温側熱交換器６１の両方から、第１高温側冷媒循環回路５０と第２高温側冷媒循環回路６０に冷媒の供給が行われることになる。また第１高温側熱交換器５１と第２高温側熱交換器６１の両方に対し、第１高温側冷媒循環回路５０と第２高温側冷媒循環回路６０から冷媒が還流することになる。

この構成によれば、第１高温側冷媒循環回路５０と第２高温側冷媒循環回路６０を、第１高温側熱交換器５１と第２高温側熱交換器６１のそれぞれに対して互いに並列に接続するから、第１高温側熱交換器５１と第２高温側熱交換器

61のいずれを取り上げても高温側冷媒循環回路が複数個確保されることになる。このため、回路が使用不可となって冷媒循環が停止し、その結果スターリング冷凍エンジン30が放熱不良でダメージを被るといった事態の回避が容易である。

加えて、第１高温側熱交換器５１と第２高温側熱交換器６１の両方において第1高温側冷媒循環回路５０と第２高温側冷媒循環回路６０に対し冷媒の供給及び還流が行われるから、第１高温側熱交換器５１と第２高温側熱交換器６１を両方とも外部への熱供給と外部からの冷熱回収に関与させることができる。

本発明冷却庫の第６実施形態を図１４に示す。第６実施形態は、第４実施形態の構成を次のように変更したものである。すなわち第４実施形態では単一型の高温側熱交換器７１を用いたが、第６実施形態では分割型の高温側熱交換器、すなわち第１高温側熱交換器５１と第２高温側熱交換器６１が用いられている。

図１４に見られるように、第１高温側冷媒循環回路５０の冷媒配管は第１高温側熱交換器５１と第２高温側熱交換器６１の両方から並列に出、途中で合流して放熱用熱交換器５２に入る。放熱用熱交換器５２を出た冷媒配管は途中で分岐し、並列をなして第１高温側熱交換器５１と第２高温側熱交換器６１に戻る。

第２高温側冷媒循環回路６０の冷媒配管は第１高温側熱交換器５１と第２高温側熱交換器６１の両方から並列に出、途中で合流して循環ポンプ64に入る。ドレンの蒸発促進のための熱交換部６２を経た後、冷却庫壁の結露防止のための熱交換部６３を出た冷媒配管は途中で分岐し、並列をなして第１高温側熱交換器５１と第２高温側熱交換器６１に戻る。

別の言い方をすれば、第１高温側冷媒循環回路５０と第２高温側冷媒循環回路６０とは、第１高温側熱交換器５１と第２高温側熱交換器６１のそれぞれに対して、互いに並列に接続されている。

上記構成により、第１高温側熱交換器５１と第２高温側熱交換器６１の両方から、第１高温側冷媒循環回路５０と第２高温側冷媒循環回路６０に冷媒の供給が行われることになる。また第１高温側熱交換器５１と第２高温側熱交換器

６１の両方に対し、第１高温側冷媒循環回路５０と第２高温側冷媒循環回路６０から冷媒が還流することになる。

以上、本発明の各実施形態につき説明したが、本発明の範囲はこれに限定されるものではなく、発明の主旨を逸脱しない範囲で種々の変更を加えて実施することができる。

産業上の利用可能性

本発明は家庭用又は業務用の冷却庫であって、スターリング冷凍機を冷熱源とするもの全般に利用可能である。

請求の範囲

１．　（補正後）スターリング冷凍エンジンにより庫内冷却を行う冷却庫において、

前記スターリング冷凍エンジンの高温部に設けた高温側熱交換器と、庫外環境に放熱を行うための放熱用熱交換器とを備え、ループ状サーモサイフォンを構成してなる第１高温側冷媒循環回路と、

前記高温側熱交換器と、ドレンの蒸発促進と冷却庫壁の結露防止の少なくとも一方に利用する熱交換器と、循環ポンプとを備え、前記高温側熱交換器の下部から該高温側熱交換器内の冷媒を前記熱交換器に送り出すように構成してなる第２高温側冷媒循環回路と、を備える冷却庫。

２．　（補正後）前記高温側熱交換器を２個設けると共に、前記第１高温側冷媒循環回路と前記第２高温側冷媒循環回路を、前記２個の高温側熱交換器のそれぞれに対して互いに並列に接続する請求項１に記載の冷却庫。

３．　（補正後）ドレンの蒸発促進のために設けられる熱交換器と、冷却庫壁の結露防止のために設けられる熱交換器とを並列接続し、それぞれの熱交換器に弁を設けて前記第２高温側冷媒循環回路を形成する請求項１または請求項２に記載の冷却庫。

４．　（補正後）前記高温側熱交換器内の冷媒は気液二相である請求項１～３のいずれか１項に記載の冷却庫。

５．　（削除）

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functions of the cold stocker, while reducing the power consumption thereof.

To achieve the above described objects, according to one aspect of the present invention, a cold stocker is structured as described below. A cold stocker that uses a Stirling refrigerating engine to cool a compartment thereof is provided with: a first warm-side refrigerant circulation circuit that is built as a loop thermosyphon and that has a warm-side heat exchanger provided in a warm section of the Stirling refrigerating engine and a heat-dissipating heat exchanger for dissipating heat to an outside environment of the cold stocker; and a second warm-side refrigerant circulation circuit that has the warm-side heat exchanger, a heat exchanger used for at least one of tasks of promoting evaporation in drainage and preventing dew condensation on a cold stocker wall, and a circulation pump. Here, the warm-side heat exchanger and the heat exchanger are so structured that a refrigerant in the warm-side heat exchanger is sent out to the heat exchanger.

With this structure, the first warm-side refrigerant circulation circuit is provided for dissipating heat of the warm section of the Stirling refrigerating engine to outside, and thus heat of the warm section can be dissipated stably. Furthermore, since the first warm-side refrigerant circulation circuit is built as a loop thermosyphon between the warm-side heat exchanger and the heat-dissipating heat exchanger, it is possible to obtain heat from the warm-side heat exchanger without using artificial energy.

In the second warm-side refrigerant circulation circuit, from the bottom of the warm-side heat exchanger, the refrigerant in the warm-side heat exchanger is pumped out into the heat exchanger by the circulation pump, and thus the heat of the warm section can surely be used for at least one of tasks of promoting evaporation in drainage and preventing dew condensation on the cold stocker wall. This makes it possible to achieve maintenance-free drainage. Furthermore, dew condensation on the cold stocker wall can be prevented without

using an electric heater, and this helps improve the performance and the user-friendliness of the cold stocker, and keeps the power consumption lower than in a case where an electric heater is used for heating.

Furthermore, from drained water and part where there is concern of dew condensation, cold having a temperature lower than the ambient temperature is collected and used to cool the warm section of the Stirling refrigerating engine, and this helps improve the heat dissipation efficiency of the whole heat dissipation system. The COP of the Stirling refrigerating engine is also improved so as to reduce the power consumption of the cold stocker.

Furthermore, the circulation pump in the second warm-side refrigerant circulation circuit does not need to be constantly operated but needs to be operated only when promoting evaporation in drainage or preventing dew condensation around a door is necessary. This helps reduce the power consumption of the circulation pump and prolong the operational lifetime thereof. In addition, the part around the door is not heated for an unnecessarily long time, and thus thermal load of the cold stocker can be reduced so as to reduce the power consumption thereof.

According to the present invention, in the cold stocker structured as just described, two of the warm-side heat exchangers are provided, and also, the first warm-side refrigerant circulation circuit and the second warm-side refrigerant circulation circuit are connected in parallel with each of the two warm-side heat exchangers.

With this structure, since two of the warm-side heat exchangers are provided, and also since the first warm-side refrigerant circulation circuit and the second warm-side refrigerant circulation circuit are connected in parallel with both of the two warm-side heat exchangers, a plurality of warm-side refrigerant circulation circuits are ensured with respect to

each of the warm-side heat exchangers. This makes it easier to avoid a situation where the circulation of refrigerant is suspended because of a circuit blockage.

Furthermore, from both of the two warm-side heat exchangers, the refrigerant is supplied to the first and second warm-side refrigerant circulation circuits, and to both of the two warm-side heat exchangers, the refrigerant flows back from the first and second warm-side refrigerant circulation circuits. This makes both of the two warm-side heat exchangers involved in supplying heat to outside.

According to the present invention, in the cold stocker structured as just described, a heat exchanger provided for promoting evaporation in drainage and a heat exchanger provided for preventing dew condensation on the cold stocker wall are connected in parallel with each other and are provided with a valve one for each, so as to form the second warm-side refrigerant circulation circuit.

With this structure, since the heat exchanger provided for promoting evaporation in drainage and the heat exchanger provided for preventing dew condensation on the cold stocker wall are connected in parallel with each other, the flow resistance of the refrigerant is kept low. Since the flow resistance of the refrigerant is low, when the circulation pump is used, the power consumption thereof is greatly reduced. In addition, since each of the heat exchangers is provided with a valve, it is possible to prevent the refrigerant from flowing in one of the heat exchangers in which the refrigerant does not need to flow at the moment so as to reduce the load on the circulation pump, and thereby to reduce the power consumption of the circulation pump.

Furthermore, according to the present invention, in the cold stocker structured as just described, the refrigerant in the warm-side heat exchanger is in a gas-liquid two-phase condition.

With this structure, since the refrigerant is used in a gas-liquid two-phase condition, latent heat can be used, through evaporation and condensation of the refrigerant, to realize heat exchange. This helps keep thermal resistance low, and thereby improves the heat dissipation efficiency. Hence, the heat exchange efficiency is significantly improved, the efficiency of the Stirling refrigerating engine is improved, and the power consumption is reduced.

Brief description of drawings

FIG. 1 is a sectional view of a cold stocker.

FIG. 2 is a piping arrangement diagram showing a cold stocker according to a first embodiment of the present invention.

FIG. 3 is a piping arrangement diagram showing a cold stocker according to a second embodiment of the present invention.

FIG. 8 is a piping arrangement diagram showing a cold stocker according to a third embodiment of the present invention.

FIG. 9 is a piping arrangement diagram showing a cold stocker according to a fourth embodiment of the present invention.

FIG. 13 is a piping arrangement diagram showing a cold stocker according to a fifth embodiment of the present invention.

FIG. 14 is a piping arrangement diagram showing a cold stocker according to a sixth embodiment of the present invention.

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Best mode for carrying out the invention

Hereinafter, embodiments of the present invention will be explained with reference to the accompanying drawings.

FIG. 1 is a sectional view showing a cold stocker. A cold stocker 1 is for preserving food, and is provided with a housing 10 having a thermal insulation structure. The housing 10 is provided with three cooling compartments 11, 12, and 13 formed one over another. The cooling compartments 11, 12, and 13 have openings, one each, on the front side of the housing 10 (in FIG. 1, on the left side), and these openings are closed with thermal insulation doors 14, 15, and 16, respectively, that are fitted freely openable and closable. On the back face of the thermal insulation doors 14, 15, and 16, gaskets 17 are attached, one each, so as to enclose the openings of the cooling compartments 11, 12, and 13, respectively, when the thermal insulation doors are shut. Inside the cooling compartments 11, 12, and 13, a shelf suitable for the type of food stored therein is arranged as necessary.

From a top, to a rear, and further to a bottom of the housing 10, a cooling system and a heat dissipation system are arranged having a Stirling refrigerating engine as their main component. FIG. 1 (sectional view) and FIG. 2 (piping arrangement diagram) show a first embodiment thereof.

In a corner between the top and the rear of the housing 10, a mounting space 19 is formed, in which a Stirling refrigerating engine 30 is mounted. Part of the Stirling refrigerating engine 30 is a cold section, to which a cold-side heat exchanger 41 is fitted. In the back of the cooling compartment 13, a compartment-cooling heat exchanger 42 is mounted. The cold-side heat exchanger 41 and the compartment-cooling heat exchanger 42 are connected to each other via a refrigerant pipe so as to form a cold-side refrigerant circulation circuit 40 (see FIG. 2). The cold-side refrigerant circulation circuit 40 is charged with a natural refrigerant such as CO_2. Inside the cold-side heat exchanger 41, a large number of fins are arranged, and this makes it possible to achieve efficient heat exchange between the refrigerant and the cold-side heat exchanger 41.

Inside the housing 10, there is provided a duct 20 for distributing to the cooling compartments 11, 12, and 13 air from which heat has been absorbed by the compartment-cooling heat exchanger 42. In the duct 20, there are properly located cold air outlets that communicate with the cooling compartments 11, 12, and 13. Inside the duct 20, there are properly located blower fans 22 for forcibly sending cold air to the cooling compartments 11, 12, and 13.

The housing 10 is also provided with a duct, which is not illustrated, for collecting air from the cooling compartments 11, 12, and 13. This duct has an air outlet below the compartment-cooling heat exchanger 42, and supplies the compartment-cooling heat exchanger 42 with air to be cooled as indicated by the dotted line arrow in FIG. 1.

Below the compartment-cooling heat exchanger 42, a drain chute 25 is arranged. The drain chute 25 collects drain that drips from the compartment-cooling heat exchanger 42, and permits the collected drain to flow into a drain pan 26.

Another part of the Stirling refrigerating engine 30 is a warm section, to which a warm-side heat exchanger is fitted. In the first embodiment, the warm-side heat exchanger is composed of a first warm-side heat exchanger 51 and a second warm-side heat exchanger 61, both of which are half-ring shaped. Inside both the first warm-side heat exchangers 51 and the second warm-side heat exchanger 61, a large number of fins are arranged, and this makes it possible to achieve efficient heat exchange between the refrigerant and the first and second warm-side heat exchangers 51 and 61.

If the warm-side heat exchanger is whole-ring-shaped, in order to fit it firmly to the warm section of the Stirling refrigerating engine 30, a strict shape control is required so as to obtain sufficient fitting accuracy. In this embodiment, by contrast, since the first warm-side heat exchanger 51 and the second warm-side heat exchanger 61 are half-ring shaped, it is

possible to control the contact pressure between the warm section and them by controlling the fastening pressure when they are fastened with the warm section of the Stirling refrigerating engine in between. This reduces the chance of a situation where an insufficient contact pressure resulting from a dimensional tolerance causes the heat transfer coefficient to decrease. The same is true in a case where the warm-side heat exchanger is divided into more blocks of a ring.

A first warm-side refrigerant circulation circuit 50 is built so as to include the first warm-side heat exchanger 51, and a second warm-side refrigerant circulation circuit 60 is built so as to include the second warm-side heat exchanger 61. The first warm-side refrigerant circulation circuit 50 is composed of the first warm-side heat exchanger 51, a heat-dissipating heat exchanger 52 arranged on the top of the housing 10, and a refrigerant pipe that connects these so as to form a closed loop. The heat-dissipating heat exchanger 52 is for dissipating heat into the environment outside the cold stocker, and is provided with a blower fan 53. The first warm-side refrigerant circulation circuit 50 is charged with water (which may be a water solution) or a hydrocarbon refrigerant. The first warm-side refrigerant circulation circuit 50 functions as a loop thermosyphon, and permits the refrigerant to circulate naturally.

The second warm-side refrigerant circulation circuit 60 is composed of the second warm-side heat exchanger 61, heat exchange portions 62 and 63, a circulation pump 64 for forcibly circulating the refrigerant, and a refrigerant pipe that connects these. The second warm-side refrigerant circulation circuit 60 is charged with a natural refrigerant such as water.

Incidentally, in this specification, of the second warm-side heat exchanger 61, the side from which the refrigerant is discharged is referred to as "the most upstream part" of the second warm-side refrigerant circuit 60. The circulation pump 64 is arranged at this most

upstream part.

Part of the pipe is formed in zigzags so as to serve as the heat exchange portion 62, which is arranged below the drain pan 24 for the purpose of heating the drain collected in the drain pan 24 with the heat of the refrigerant so as to promote evaporation thereof.

Part of the pipe is extended so as to run around the openings of the cooling compartments 11, 12, and 13 so as to serve as the heat exchange portion 63, which heats that part with the heat of the refrigerant in order to prevent dew condensation there.

Next, how a cold stocker 1 operates will be described.

When the Stirling refrigerating engine 30 starts to be driven, the cold section thereof is cooled and the temperature of the warm section thereon rises. Heat is absorbed from the cold-side heat exchanger 41, and the refrigerant in the cold-side heat exchanger 41 is condensed and flows through the cold-side refrigerant circulation circuit 40 into the compartment-cooling heat exchanger 42.

The refrigerant that has flowed into the compartment-cooling heat exchanger 42 evaporates in the compartment-cooling heat exchanger 42, lowering the surface temperature of the compartment-cooling heat exchanger 42. The air that flows through the compartment-cooling heat exchanger 42 is deprived of heat so as to become cold air, blows out from the cold air outlet in the duct 20 into the cooling compartments 11, 12, and 13, and lowers the temperatures in the cooling compartment 11, 12, and 13. Thereafter, the air flows though the unillustrated duct and flows back to the compartment-cooling heat exchanger 42.

The evaporated refrigerant flows through the cold-side refrigerant circulation circuit 40 and flows back to the cold-side heat exchanger 41, and is deprived of heat so as to be condensed. Then, the condensed refrigerant again flows to the compartment-cooling heat exchanger 42.

The heat generated by the Stirling refrigerating engine 30 operating is dissipated from the warm section, and so is the heat that the cold section has collected from the inside of the cooling compartments. This heat heats the first warm-side heat exchanger 51 and the second warm-side heat exchanger 61.

When the first warm-side heat exchanger is heated, the refrigerant inside thereof evaporates, and flows into the heat-dissipating heat exchanger 52. The blower fan 53 blows air to the surface of the heat-dissipating heat exchanger 52, and thus heat is absorbed from the refrigerant inside and the refrigerant becomes condensed. The condensed refrigerant flows back to the first warm-side heat exchanger 51, and evaporates again. In this way, the cycle is repeated in which the refrigerant receives heat from the warm section of the Stirling refrigerating engine 30 so as to evaporate and then gives the heat to cooling air at the heat-dissipating heat exchanger 52 so as to be condensed.

In the first warm-side refrigerant circulation circuit 50, the refrigerant is used in a two-phase condition where the gas phase and the liquid phase coexist. In heat exchange accompanied by phase changes between vapor and liquid, latent heat is exploited through evaporation and condensation of a refrigerant. This makes it possible to significantly improve heat transfer coefficient, compared with in heat exchange which is not accompanied by phase changes.

What is just described will be explained. The value of the amount of heat Q dissipated from the Stirling refrigerating engine 30 is given by the following formula:

$$Q = h\,A / \Delta\,Tm$$

where

h is heat transfer coefficient;

A is heat transfer area; and

Δ Tm is temperature difference.

Accordingly, the higher the heat transfer coefficient is, the lower the temperature of the warm section of the Stirling refrigerating engine 30 can be made, resulting in an enhanced COP.

In general, when a refrigerant is used in a brine method which is not accompanied by phase changes, the heat transfer coefficient is in the range of from several hundred to a thousand w/m2k. Furthermore, the heat transfer coefficient is proportional to the power consumption of a pump for circulating brine.

In contrast, in the heat exchange accompanied by phase changes between vapor and liquid, in which latent heat is exploited through evaporation and condensation of a refrigerant, it is possible to obtain a heat transfer coefficient in the range of 3000 to 10000 w/m2k. The value of the heat transfer coefficient is from several times to ten and several times larger than that in a brine method.

In the first warm-side refrigerant circulation circuit 50, the refrigerant is circulated in a gas-liquid two-phase condition as described above, and thus heat exchange can be carried out efficiently. The thermal resistance that arises during heat exchange is extremely low, and thus under similar conditions (similar ambient temperature, similar amount of dissipated heat), the temperature of the warm section of the Stirling refrigerating engine 30 is kept lower. Hence, the Stirling refrigerating engine 30 operates with an enhanced COP, and thus the power consumption is reduced.

When the second warm-side heat exchanger 61 is heated, the refrigerant evaporates. Also here, the refrigerant is used in a gas-liquid two-phase condition. The refrigerant in a gas-liquid two-phase condition is pumped into the heat exchange portions 62 and 63 by the circulation pump 64.

The refrigerant first flows through the heat exchange portion 62, and transfers heat to the drain pan 26 located above it. Hence, the temperature of the drain in the drain pan 26 rises without being heated with an electric heater, and thus evaporation of the drain is promoted. This eliminates the need to empty the drain pan 26 of the drain collected therein, and this makes it possible to achieve maintenance-free drainage.

Subsequently, the refrigerant flows through the heat exchange portion 63 so as to heat the vicinities of the openings of the cooling compartments 11, 12, and 13. Dew is liable to be condensed around where the gaskets 17 come in contact with the housing 10, that is, the boundary area between the inside and the outside of the cooling compartments. By permitting the refrigerant to flow, however, the temperature of the places of the cold stocker wall exposed to the ambient air is kept higher than the dew-point temperature, and thus dew condensation can be prevented without using an electric heater.

The refrigerant collects cold from drain at the heat exchange portion 62, and collects cold from the housing 10 at the heat exchange portion 63. After collecting cold in this way, the refrigerant which has been in the gas phase converts to the liquid phase, and flows into the second warm-side heat exchanger 61 in a single phase, that is, the liquid phase. There, the refrigerant in the liquid phase comes in contact with the refrigerant in the gas phase and converts it into the liquid phase so as to lower the vapor pressure. Thus, evaporation of the refrigerant in the liquid phase is promoted and a gas-liquid two-phase condition of the refrigerant is restored. In this way, a cycle is repeated in which the refrigerant receives heat from the warm section of the Stirling refrigerating engine 30 so as to evaporate and then, at the heat exchange portions 62 and 63, dissipates the heat so as to be condensed so as to collect cold. When the circulation pump 64 stops its operation, this cycle is suspended.

The refrigerant supplies heat to drain and the vicinities of the openings of the cooling

compartments 11, 12, and 13, and in exchange therefor, collects cold having a temperature lower than the ambient temperature so as to cool the warm section of the Stirling refrigerating engine 30 therewith. This reduces the thermal load on the heat dissipation system, and thus improves the heat dissipation efficiency of the whole heat dissipation system. Hence, the Stirling refrigerating engine can be operated with an enhanced COP so as to reduce the power consumption.

The first warm-side refrigerant circulation circuit 50 and the second warm-side refrigerant circulation circuit 60 are designed to be independent of each other, and are arranged in parallel with each other. This makes it possible for the first and second warm-side refrigerant circulation circuits 50 and 60 to carry out heat dissipation without depending on each other. This means that flexible control of operation modes is possible based on the thermal load condition of the cold stocker 1. For example, instead of operating the circulation pump 64 continuously, it is possible to operate it only when promotion of evaporation in drainage or prevention of dew condensation around the door is needed. This makes it possible to reduce the power consumption of the circulation pump 64 and to prolong the operational lifetime thereof.

Furthermore, since the circulation pump 64 is arranged at the most upstream part of the second warm-side circulation circuit 60, the pipe resistance from the second warm-side heat resistance 61 through the circulation pump 64 is low, and this permits the refrigerant to smoothly flow into the circulation pump 64. A large resistance in the pipe through which the refrigerant is supplied to the circulation pump 64 may cause cavitation on the inlet side of the circulation pump 64 to allow the refrigerant to evaporate unnecessarily, and result in poor circulation efficiency. Arranging the circulation pump at the most upstream part of the second warm-side refrigerant circulation circuit 60, however, helps avoid such a situation.

In regard to a gas-liquid two-phase condition of the refrigerant, in the second warm refrigerant circulation circuit 60, at the heat exchange portions 62 and 63, around where drain processing and prevention of dew condensation is performed, the refrigerant may exist solely in the liquid phase. When the refrigerant solely in the liquid phase flows back to the second warm-side heat exchanger 61, latent heat exchange takes place between the returning liquid refrigerant and the refrigerant vapor, and thus highly efficient heat exchange can be achieved here.

Next, a second and further embodiments will be described with reference to FIG. 3 and the following drawings. FIGS. 3 to 14 are piping arrangement diagrams, and the piping arrangements illustrated therein are assumed to be realized in the cold stocker 1 shown in FIG. 1. Such components as find their counterparts in the first embodiment are identified with common reference numerals, and overlapping descriptions will not be repeated.

The second embodiment of the cold stocker of the present invention is illustrated in FIG. 3. Here, the heat exchange portion 62 for promoting evaporation in drainage and the heat exchange portion 63 for preventing dew condensation on the cold stocker wall are connected in parallel with each other, and this parallel connection configuration is connected in series with the second warm-side heat exchanger 61 and the circulation pump 64. In this embodiment, too, the circulation pump 64 is arranged at the most upstream part of the second warm-side refrigerant circulation circuit 60. Inside the parallel connection configuration, a valve 65 is connected to the heat exchange portion 62 at the upstream side thereof, and a valve 66 is connected in series with the heat exchange portion 63 at the upstream side thereof.

With the above structure, the flow resistance of the refrigerant at the heat exchange portions 62 and 63 is approximately half that in the first embodiment, and thus the power consumption of the circulation pump 64 is reduced significantly. Furthermore, since the

valves 65 and 66 are connected with the heat exchange portions 62 and 63, respectively, if whichever of promotion of evaporation in drainage and prevention of dew condensation on the cold stocker wall is not needed, whichever of the valve not needed may be shut so as to stop the refrigerant from flowing therethrough. This reduces the load on the circulation pump, and helps further reduce the power consumption of the circulation pump 64.

The valve 66 may be kept closed unless necessary to prevent condensation. This prevents the part around the doors 14, 15, and 16 from being heated longer than necessary. In this way, it is possible to reduce the thermal load on the cooling compartments 11, 12, and 13, and thereby save power consumption.

Instead of two valves dedicated to the heat exchange portions 62 and 63, respectively, a three-way valve may be shared that is switched to select one of the following three states: the refrigerant flowing through both of the heat exchange portions 62 and 63; the refrigerant flowing through only the heat exchange portion 62; and the refrigerant flowing through only the heat exchange portion 63. In order to achieve easy automatic control, it is preferable that the valve be a solenoid valve.

Incidentally, the refrigerant that flows through the first warm-side refrigerant circulation circuit 50 is in a gas-liquid two-phase condition, and so is the refrigerant that flows through the second warm-side refrigerant circulation circuit 60.

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A third embodiment of the cold stocker of the present invention is illustrated in FIG. 8. The third embodiment is identical to the second embodiment except for that the warm-side heat exchanger thereof is built as one block. That is, in this embodiment, the warm-side heat exchanger 71 made of one block is mounted to the warm part of the Stirling engine 30. Inside the warm-side heat exchanger 71, a large number of fins are provided so as to achieve efficient heat exchange with the refrigerant.

The first warm-side refrigerant circulation circuit 50 and the second warm-side refrigerant circulation circuit 60 are formed to include the warm-side heat exchanger 71. That is, the warm-side heat exchanger 71 is a warm-side heat exchanger shared by the first warm-side refrigerant circulation circuit 50 and the second warm-side refrigerant circulation circuit 60, and both of the first warm-side refrigerant circulation circuit 50 and the second warm-side circulation circuit 60 are connected in parallel with this shared warm-side heat

exchanger 71.

A fourth embodiment of the cold stocker of the present invention is illustrated in FIG. 9. In a humid environment, promotion of evaporation in drainage and prevention of dew condensation on the cold stocker wall need to be carried out continuously, and the piping arrangement of the fourth embodiment is suitable for such a case.

The fourth embodiment is identical to the first embodiment except for that the warm-side heat exchanger thereof is built as one block. That is, in this embodiment, the warm-side heat exchanger 71 made of one block is mounted to the warm section of the Stirling engine 30. Inside the warm-side heat exchanger 71, a large number of fins are arranged so as to achieve efficient heat exchange with the refrigerant.

The first warm-side refrigerant circulation circuit 50 and the second warm-side refrigerant circulation circuit 60 are formed to include the warm-side heat exchanger 71. That is, the warm-side heat exchanger 71 is a warm-side heat exchanger shared by the first warm-side refrigerant circulation circuit 50 and the second warm-side refrigerant circulation circuit 60, and both of the first warm-side refrigerant circulation circuit 50 and the second warm-side circulation circuit 60 are connected in parallel with this shared warm-side heat exchanger 71.

With the above structure, the piping arrangement in the warm-side refrigerant circulation circuit 60 can be advantageously simple and the number of steps in the assembly process can be advantageously reduced.

The arrangement of the heat exchange portions 62 and 63 may be reversed, that is, they may be so located that the cold stocker wall is first heated and then the drain pan 26 is heated.

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A fifth embodiment of the cold stocker of the present invention is illustrated in FIG. 13. The fifth embodiment is structured by modifying the second embodiment as below. In the second embodiment, the first warm-side heat exchanger 51 is dedicated to the first warm-side refrigerant circulation circuit 50, and the second warm-side heat exchanger 61 is dedicated to the second warm-side refrigerant circulation circuit 60. In the fifth embodiment, the first warm-side refrigerant circulation circuit and the second warm-side refrigerant circulation circuit share both of the first and second warm-side heat exchangers 51 and 61.

As shown in FIG. 13, in the first warm-side refrigerant circulation circuit 50, two refrigerant pipes come out in parallel with each other, one from each of the first and second warm-side heat exchangers 51 and 61, are joined together so as to be a single pipe along the way, and then enters the heat-dissipating heat exchanger 52. The refrigerant pipe coming out of the heat-dissipating heat exchanger 52 is split into two parallel pipes along the way, so that each, in parallel with each other, enters back the first warm-side heat exchanger 51 and

the second warm-side heat exchanger 61.

In the second warm-side refrigerant circulation circuit 60, a refrigerant pipe comes out, from each of the first and second warm-side heat exchangers 51 and 61, in parallel with each other, are joined together so as to be a single pipe along the way, and enters the circulation pipe 64. The refrigerant pipe coming out from the parallel connection structure of the heat exchange portion 62 for promoting evaporation in drainage and from the heat exchange portion 63 for preventing dew condensation on the cold stocker wall is split along the way so as to enter back, in parallel with each other, the first and second warm-side heat exchangers 51 and 61.

In other words, the first warm-side refrigerant circulation circuit 50 is connected in parallel with the first warm-side heat exchanger 51 and in parallel with the second warm-side heat exchanger 61; and the second warm-side refrigerant circulation circuit 60 is connected in parallel with the first warm-side heat exchanger 51 and in parallel with the second warm-side heat exchanger 61.

With the above structure, from both the first and second warm-side heat exchangers 51 and 61, the refrigerant is supplied to the first and second warm-side refrigerant circulation circuits 50 and 60. Furthermore, into both the first and second warm-side heat exchangers 51 and 61, the refrigerant flows back from the first and second warm-side refrigerant circulation circuits 50 and 60.

With this structure, the first warm-side refrigerant circulation circuit 50 is connected in parallel with the first warm-side heat exchanger 51 and in parallel with the second warm-side heat exchanger 61, and the second warm-side refrigerant circulation circuit 60 is connected in parallel with the first warm-side heat exchanger 51 and in parallel with the second warm-side heat exchanger 61. Thus, with respect to each of the first and second

warm-side heat exchangers, a plurality of warm-side refrigerant circulation circuits can be ensured. Hence, a situation can be easily avoided where a circuit becomes unusable preventing the refrigerant from circulating, resulting in the Stirling refrigerating engine 30 being damaged by insufficient heat dissipation.

In addition, since both in the first and second warm-side heat exchangers 51 and 61, the refrigerant is supplied to and flows back from the first and second warm-side refrigerant circulation circuits 50 and 60, both the first and second warm-side heat exchangers 51 and 61 can be included in dissipating heat to outside and collecting cold from outside.

A sixth embodiment of the cold stocker of the present invention is illustrated in FIG. 14. The sixth embodiment is structured by modifying the fourth embodiment as below. That is, in the fourth embodiment, the warm-side heat exchanger 71 is built as one block, but in the sixth embodiment, a warm-side heat exchanger is separated into two, that is, the first warm-side heat exchanger 51 and the second warm-side heat exchanger 61 are used.

As shown in FIG. 14, in the first warm-side refrigerant circulation circuit 50, from each of the first and second warm-side heat exchangers 51 and 61, a refrigerant pipe comes out in parallel with each other, and are joined together into one refrigerant pipe so as to enter the heat-dissipating heat exchanger 52. The refrigerant pipe coming out of the heat-dissipating heat exchanger 52 is separated into two parallel pipes along the way so that each, in parallel with each other, enters back the first warm-side heat exchanger 51 and the second warm-side heat exchanger 61.

In the second warm-side refrigerant circulation circuit 60, a refrigerant pipe comes out, from each of the first and second warm-side heat exchangers 51 and 61, in parallel with each other, are joined together so as to be a single pipe along the way, and enters the circulation pump 64. The refrigerant pipe coming out from the heat exchange portion 62 for

promoting evaporation in drainage and then from the heat exchange portion 63 for preventing dew condensation on the cold stocker wall is split along the way so as to enter back, in parallel with each other, the first and second warm-side heat exchangers 51 and 61.

In other words, the first warm-side refrigerant circulation circuit 50 is connected in parallel with the first warm-side heat exchanger 51 and in parallel with the second warm-side heat exchanger 61, and the second warm-side refrigerant circulation circuit 60 is connected in parallel with the first warm-side heat exchanger 51 and in parallel with the second warm-side heat exchanger 61.

With the above structure, from both the first and second warm-side heat exchangers 51 and 61, the refrigerant is supplied to the first and second warm-side refrigerant circulation circuits 50 and 60. Furthermore, into both the first and second warm-side heat exchangers 51 and 61, the refrigerant flows back from the first and second warm-side refrigerant circulation circuits 50 and 60.

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Embodiments of the present invention have been explained above, but it should be understood that they are not meant to limit the application of the present invention in any manner, and that various modifications are permissible within the spirit of the present invention.

Industrial applicability

The present invention is a cold stocker for household use or for business use, and is applicable to appliances in general that use a Stirling refrigerating engine as their cold source.

CLAIMS

1. (Amended) A cold stocker that uses a Stirling refrigerating engine to cool a compartment thereof, comprising:

a first warm-side refrigerant circulation circuit that is built as a thermosyphon and that comprises a warm-side heat exchanger provided in a warm section of the Stirling refrigerating engine and a heat-dissipating heat exchanger for dissipating heat to an outside environment of the cold stocker; and

a second warm-side refrigerant circulation circuit that comprises: the warm-side heat exchanger; a heat exchanger used for at least one of tasks of promoting evaporation in drainage and preventing dew condensation on a cold stocker wall; and a circulation pump,

wherein the warm-side heat exchanger and the heat exchanger are so structured that a refrigerant in the warm-side heat exchanger is sent out to the heat exchanger.

2. (Amended) The cold stocker of claim 1, comprising two of the warm-side heat exchangers, wherein

the first warm-side refrigerant circulation circuit and the second warm-side refrigerant circulation circuit are connected in parallel with each of the two warm-side heat exchangers.

3. (Amended) The cold stocker of claim 1 or claim 2, wherein

a heat exchanger provided for promoting evaporation in drainage and a heat exchanger provided for preventing dew condensation on the cold stocker wall are connected in parallel with each other and are provided with a valve one for each, so as to form the

second warm-side refrigerant circulation circuit.

4. (Amended) The cold stocker of one of claims 1 to 3, wherein

a refrigerant in the warm-side heat exchanger is in a gas-liquid two-phase condition.

5. (Deleted)

6. (Deleted)

7. (Deleted)

8. (Deleted)

9. (Deleted)

10. (Deleted)

11. (Deleted)

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15. (Deleted)

(page 48 to 50 have been deleted)